Using Internal Vibration Transducers to Improve Fault Detection on Aeroderivative Gas Generators

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Abstract
This paper discusses a research project undertaken by Bently Nevada Corporation and TransCanada PipeLines to develop an improved machinery condition monitoring system for the Rolls-Royce Avon gas turbine engine. The project centers on the installation of internal vibration and temperature transducers in and around the engine internal wheel case housing for the center bearing. The paper outlines the current industry-accepted vibration monitoring system, why there is a need to improve the system, the design of the internal transducer mounting arrangements, and test results of the new system.

Avon General Design Overview and History
Rolls-Royce began development of the AJ65 military aircraft engine in early 1945. Its 6500-pound thrust was improved in 1953 through a modified compressor design, and resulted in the 200 series RA14 tested at 9500 pounds of thrust. Continued developments in the 300 series resulted in the RB146 which produced 17,110 pounds of thrust and was renamed Avon (Figure 1). Today's land-based primary drive version of the Avon has an installed base of approximately 1185 units throughout the world. Parts, overhaul, and Dry Low Emissions (DLE) retrofit engines are the main Avon product still supported by Rolls-Royce and other rebuild shops. The single-spool gas generator holds a 3-stage turbine driving a 17-stage compressor producing a compression ratio of approximately 10:1. The gas generator casing is made up of six major subassemblies consisting of: (1) the air inlet casing, (2) the compressor casing, (3) the compressor outlet casing, (4) the combustion chamber section, (5) the turbine casing, and (6) the exhaust unit. These casing sections bolt together to form the engine structure. Alignment of the casings relative to each other is by dowel pins and toleranced bolts. The air inlet, compressor, and internal wheel case (location of the center bearing) are manufactured of light alloys. The compressor outlet, combustion, turbine, and exhaust casings are steel. The combustion section contains eight can-type combustors.

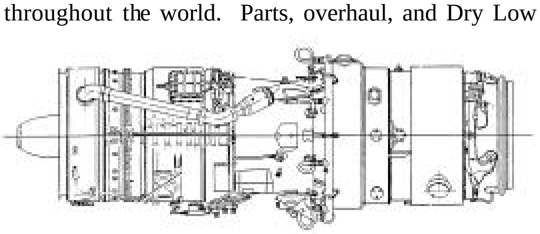


Figure 1 – Avon Gas Generator

The rotor is supported by three rolling element bearings located in the air inlet, compressor outlet, and combustion chamber sections. The front and rear bearings are cylindrical roller element bearings. The center bearing, containing 12 balls, handles both radial and axial rotor loads. Axial loading on the rotor results from turbine wheels loaded toward the aft, from pressure drop across the stages, and from compressor wheels loaded in the forward direction from the pressure increase across each stage. The net thrust loading for an aircraft engine is, of course, toward the front of the engine. Bearing lubrication is via a constant-pressure system and oil sprays.

Cooling air from a compressor intermediate stage is used to cool components not in direct contact with the main gas stream. Compressor discharge air is used to cool the turbine section and blades. Seals are labyrinth self-clearing or thread-type and are pressurized with cooling air to prevent leakage of air, oil, or hot gas into vital components.

Airflow through the compressor is controlled by 23 moveable inlet guide vanes and two bleed valves in the compressor casing. This system allows a wide range of operating speeds and modifies the optimal compressor operation curve for inlet air and desired speed conditions.

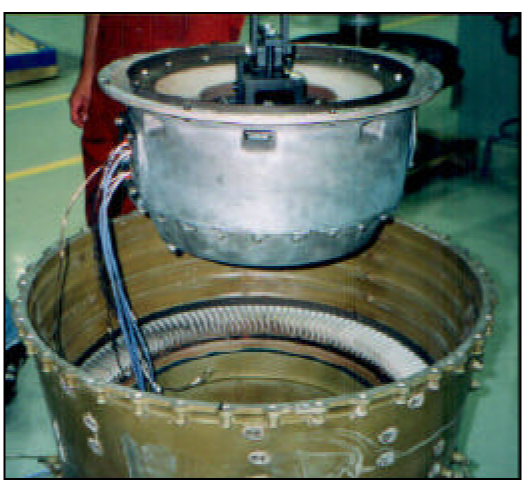
Current Monitoring Methodology
Vibration monitoring for land-based engines evolved from their aircraft cousins to warn pilots of impending engine failure. Currently, the Rolls-Royce-approved monitoring system for land-based Avons consists of two case-mounted accelerometers. One is horizontally mounted on the right side facing the GG inlet casing, and the other is horizontally mounted on the left side of the compressor discharge casing near the center bearing. Signals from these transducers are filtered to pass only those frequencies between 40 Hz and 350 Hz, and mathematically integrated to convert from acceleration to velocity units. Casing velocity in engineering units of in/sec peak, or mm/sec peak, is then monitored in a rack-mounted instrument where it is compared against pre-established alarm limits.

Such a monitoring system and filtering scheme effectively provides only vibration at shaft rotative speeds. Thus, readings from such a system are used by station operations primarily as an indicator of high rotor imbalance (generally indicative of blade damage) in order to trip the unit prior to catastrophic or excessive internal damage. The heavily filtered data provided from these transducers is of very limited diagnostic value. As such, continuous or diagnostic engine health monitoring as a way to reduce engine maintenance costs, better predict engine degradation, or extend engine reliability is significantly hampered.

For the more sophisticated end-users of the engine, some limited health diagnostics can be performed by tapping into these signals with data recording instrumentation. However, the usefulness of the data is limited by several factors: the filtering of the signal, the casing attenuation and amplification of various components within the signals, the lack of a true once-per-turn Keyphasor® signal to provide phase data, the location and number of transducers, and the type of transducer used. For these reasons, Bently Nevada, in conjunction with TransCanada PipeLines, decided to investigate an improved health monitoring system based on internally mounted bearing-observing proximity probe transducers. The goal of the system is to provide better quality engine data that will allow operators to better monitor, document, and predict engine health. Expected results include better scheduling of engine overhauls, reduced maintenance, fewer expensive bearing repairs, reduced unexpected catastrophic engine failures, and potentially better operating guidelines extending engine reliability and Mean Time Between Failure (MTBF).

Internal Transducer Installation

The research engine used for this project is owned by TransCanada PipeLines and is a Mark 1535-161G series Avon, serial #37510. The unit is used to drive a booster compressor in natural gas pipeline service and is operated continuously near 95% capacity. This particular engine had 134,461 lifetime operation hours and 14,817 hours since last overhaul when the internal transducers were installed.

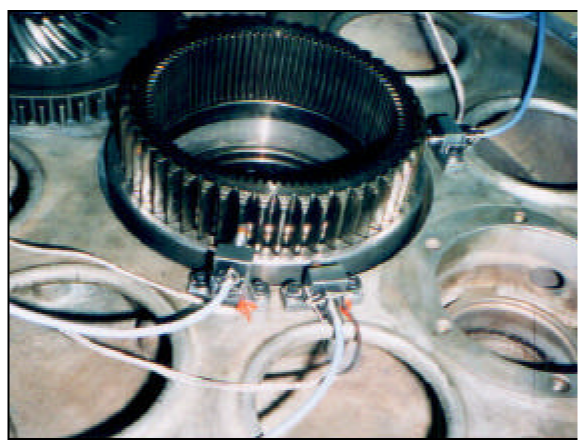


Photograph 1. Internal wheel case showing routing of transducer cables.

This unit, and in particular the center bearing, was selected as the best location for the transducer installation because of its history of center bearing failures.

The internal transducers installed in the engine were located within the internal wheel case. This casing houses the center bearing as well as gearing for the power take-off drive shaft and is located within the compressor outlet casing. As shown in Photograph 1, taken during installation of the internal wheel case in the compressor outlet casing, transducer wiring was routed through a specially designed cable seal arrangement mounted in place of a cover plate normally installed opposite the power take-off drive shaft. The wheel case is designed to allow installation of the power take-off shaft on either side of the housing for aircraft port or starboard engine mounting.

The internal wheel case is shown in Figure 2 on the following page, and diagrams of the transducer arrangements can be found in Figures 3, 4, 11, 12, and 13. Installed transducers include 3 REBAM® probes, 2 shaft probes, 1 speed probe, 1 high temperature accelerometer, and 6 resistance temperature detectors (RTDs). All proximity probes are 5 mm, high temperature versions. The two existing casing-mounted accelerometers supplied as part of the OEM-standard Avon monitoring package were left intact and used for comparison with the newly installed internal transducers.



Photograph 2. Shaft proximity probes & speed probe.

The shaft proximity probes, shown in Photograph 2 and in Figure 12, are mounted to view the shaft coupling just forward of the rear casing. This was the only available shaft location that provided a suitable target area for the probes. The axial position of the edge of the coupling varies by what appears to be as much as 2 mm to 3 mm during installation, depending on the build up of the remaining sections of the engine. In this installation, the shaft probe tips extended over the coupling edge by 1 mm to 1 ½ mm when the engine was assembled, but still provided excellent data.

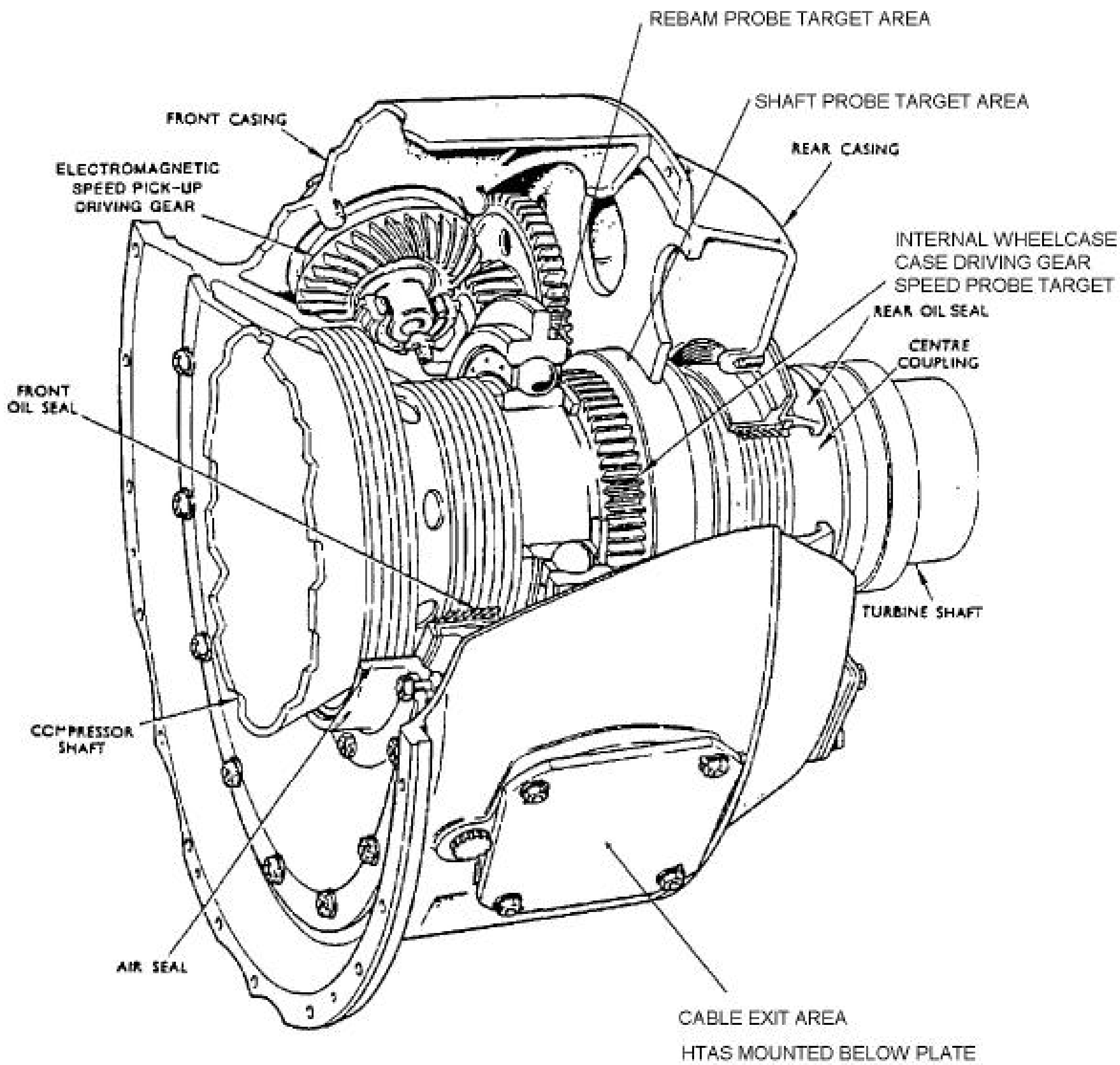
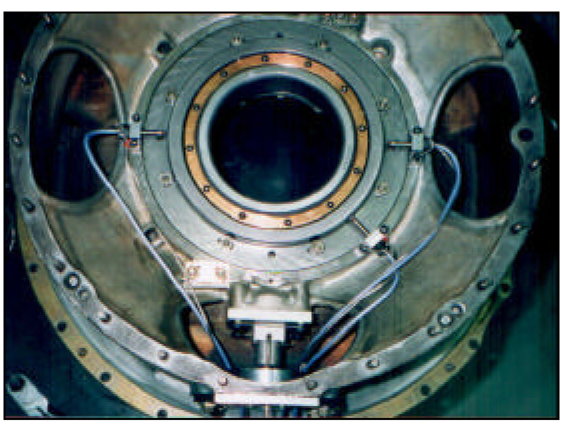


Figure 2 – Internal wheel case



Photograph 3. REBAM® probes & internal accelerometer.

The three Rolling Element Bearing Activity Monitor (REBAM®) probes, shown in Photograph 3 and Figure 11, were mounted on the forward half of the internal wheel case at the edge of the center bearing housing. The probes view the outer race of the center bearing on the aft side of the bearing bolt flange.

What Is a REBAM® Probe?

REBAM probes are eddy current proximity probes that use high-gain, low-noise Proximitor® signal conditioners to make direct measurements of the bearing outer race movement relative to the bearing housing in response to the rolling element passage.

Micro-inch deflections of the outer race transmit information regarding the condition of the bearing rollers/balls, races, and cage, and offer a high signal-to-

noise ratio compared to casing-mounted velocity transducers or accelerometers. Defects in the rolling elements or bearing races shock and deflect the outer race as they pass. These probes were selected because they detect that deflection at the earliest stages, often long before bearing degradation is significant enough to produce acceleration readings above the noise floor of an accelerometer.

Although the configuration of this bearing is less than an ideal REBAM® installation because of the bearing-stiffening flange, the transducers are providing significant amounts of useful data.

The high-temperature accelerometer installed was a specially modified Bently Nevada High Temperature Accelerometer System (HTAS). This accelerometer is mounted onto the unused power take-off (PTO) shaft bearing housing. Its axis of measurement is oriented radial to and centered over the outer race of the center bearing.

The Keyphasor® probe, originally intended to view a dimple machined on the shaft coupling (see Figures 12 and 13), was not installed because of an interference of the coupling during engine assembly. The once-per-turn signal is now obtained from the speed probe viewing the 51-tooth gear. The signal is routed through a Bently Nevada TK-16 Keyphasor Multiplier/Divider and divided by 51. This technique is not ideal because the reference point changes from run to run, but does provide a once-per-turn reference for testing.

During the overhaul at TransCanada Turbines in Calgary, a new center bearing was installed to provide baseline data for comparison. Forward and aft bearings were not replaced.

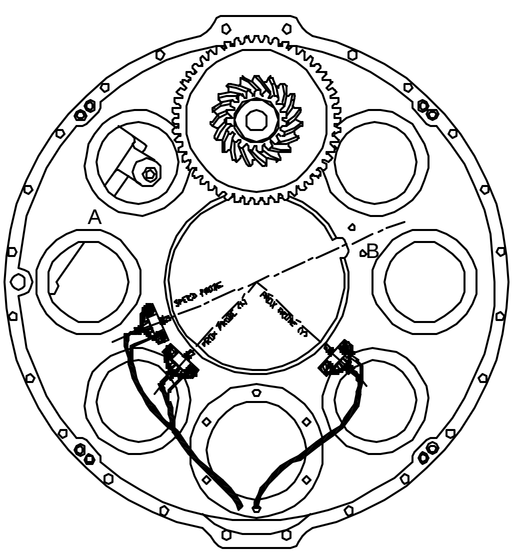


Figure 3. Aft half internal wheel case looking aft.

Hardware Installation Methodology

The design modifications for the existing wheel case hardware included tapped holes, a small cutout, and the addition of a new cover plate with threaded cable seals. These modifications were independently reviewed to ensure that none of the machining work would create unacceptable material stresses in the wheel case. Radial, tangential (hoop), and axial loads were examined to determine any debit in load-carrying capability in these directions caused by modifications to the wheel case. Both axial and radial load-carrying capabilities were unaffected by any of the modifications, as they did not fall within the critical section of thin webbing between existing holes in the aft half of the wheel case (indicated by 'A' in Figure 3).

Tangential or hoop load capability was affected by the probe bracket mounting holes. Load calculations indicated a 6% decrease in the hoop load capability of the plate because of the tapped holes near the Keyphasor® probe cutout opposite the speed probe (indicated by 'B' in Figure 3). To determine if a 6% debit in the tangential load capability was acceptable, the following Avon general design criteria were taken into consideration:

- Load on the plate is from gear mesh forces which are radial and axial, not tangential.
- Unidirectional loads can be slightly redistributed to other plate locations without consequence.
- The plate has no thermal loads or full hoop loads.
- The Avon was designed as a flight engine subject to maneuver loads and landing jolts of up to 20 G's instantaneous.
- Flight engines were designed for faster rotor accelerations, producing higher gearing loads.
- Design margins of safety for flight engines are at least 15%.
- Gear loading on the PTO shaft is well below design as the shaft only turns a sensor wheel.
- The analysis ignores the tangential capability outboard of the lightening holes.

For these reasons the 6% tangential load debit was considered acceptable.

Stress riser concentrations from the cutout and the tapped holes were considered, but are only applicable in fatigue life calculations. Fatigue has historically never been a failure mode of the rear half of the wheel case and the stress concentrations and resultant fatigue life reduction were considered academic. Prior to installation, all installed mounting hardware and machined casing elements were inspected using dye penetrate and Magnaflux® to ensure no cracks were present.

Transducer hardware is mounted in accordance with Bently Nevada best practices as shown in Figure 4. All probes are threaded into mounting blocks and secured with jam nuts. Jam nuts are secured with safety wire locking clips. Probe mounting block screws are threaded into a tapped hole in the wheel case body and safety-wired to the jam nut locking clip. Additionally, #243 Locktite® and split-lock

washers are installed on the screws. RTD's are inserted into a hole on each mounting block and secured with high temperature RTV potting compound. Probe and RTD cables are then secured with high temperature heat-shrink tubing up to the probe body.

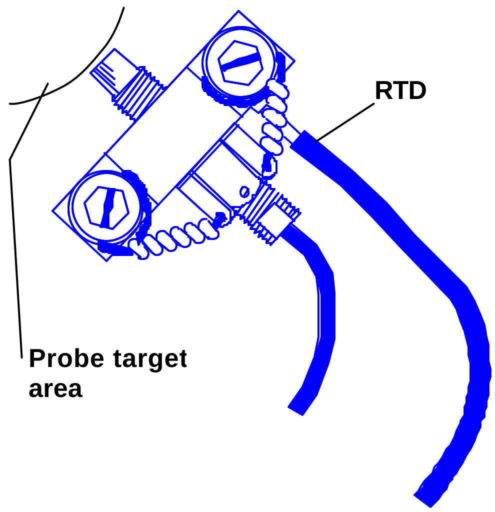


Figure 4. Proximity probe and RTD mounting scheme

The HTAS and the HTAS mounting plate is secured with safety-wired cap screws, split-lock washers, and #243 Locktite threaded into the mounting plate and the wheel case body. Wire runs within the wheel case are less than 7 inches to the low-pressure cable seal, and wire tie-downs were not used.

The cable seal is a two-piece split design to allow multiple cables and cables with a large end-connector to be installed. The seals are threaded into the cable exit plate, tightened and safety wired together. They are then injected with a high temperature RTV potting compound such as Dow #736. The RTV seals around the cable and forms an internal locking plug within the cavity of the seal. The cable exit plate is secured with safety-wired screws threaded into the body of the wheel case. All mounting hardware is fabricated from cold-rolled 1018 steel or equivalent.

Future Design Changes

Future design enhancements include:

- Installation of a once-per-turn Keyphasor® probe viewing a flat spot ground on the shaft coupling major diameter.
- Addition of a second speed probe as a redundant backup.
- Hex head cap screws with locking-tab washers to replace the safety-wired socket head cap screws securing the probe mounting blocks.
- Locking-tab washers to replace the safety wire to secure the cable seals.
- Locking-tab washers to replace the safety-wire locking clips that secure the proximity probes lock nut in the mounting blocks.
- A modified HTAS or similar high temperature accelerometer.

Data Acquisition

After re-assembly, the engine was full-power tested at TransCanada Turbine's test cell and then installed at the TransCanada PipeLines Dusty Lake compressor station. Transient and steady-state data from the test cell and from Dusty Lake was initially sampled using a Bently Nevada ADRE® system. A Bently Nevada Data Manager® 2000 system installed at the station continuously monitored the unit's speed, vibration, and temperature data after installation. In May 2000, the unit was transferred to the Smoky Lake compression station where the engine would run continuously. There, the engine transducers were connected to a Bently Nevada 3500 Series machinery protection system as well as Data Manager 2000 software to continuously monitor the engine.

The test cell and initial station data is considered baseline for a well running unit with a new center bearing and was used to evaluate future operating conditions.

Baseline Data from REBAM® Probes

The value of REBAM data in other types of machinery has historically been to trend bearing outer race deflection amplitudes as an indicator of race or ball wear problems. Typically, outer race deflection amplitudes of 10 to 20 µin peak-to-peak (pp) have been cause for concern over the condition of the bearing in most industrial applications. This new engine bearing exhibited operating amplitudes initially much higher than expected.

REBAM data waveforms contain three predominant components at running speed: Large rotor 1X amplitudes on the order of 20 to 150 µin pp; 2X components in the range of 15 to 30 µin pp; and 39 to 41 event/revolution spikes on the order of 6 to 12 µin pp during rotor operating speeds of 6800 rpm to 7800 rpm. Numerous other frequency components are present within the data, either of much smaller amplitude or of erratic duration. These are considered noise, or lower-level harmonics. Figure 5 shows the timebase and spectrum plots from the 180° REBAM transducer with the unit at 7800 rpm.

The large rotor 1X component is believed to be the result of bearing movement within the bearing housing and not outer race deflection. Eight toleranced studs evenly spaced around the flange on the bearing outer race control the fit of the bearing within the housing. These studs, through the flange, hold the bearing in the housing and take the entire rotor thrust load and some of the radial load. The major diameter of the flange contains an O-ring within a groove as an oil galley seal and is a slip fit within the housing.

The fit tolerance of the studs within the flange holes and between the outer rim and the housing is unknown, but is

suspected to be loose because of the amount of 1X amplitude observed on the bearing outer race.

Although a finite element deflection analysis of the outer race has not been undertaken, the amount of 1X amplitude is not believed to be the result of race bending, but rather movement on the studs and within the housing. Further information on the stud-to-flange hole fit and race deflection is required to determine the source(s) of this large 1X amplitude.

The 2X component appears to be proportional to the 1X amplitude and may be the result of the bearing outer race fit within the housing, a rotor asymmetrical stiffness within the shaft coupling, or internal bearing clearances. This data is still being evaluated.

The 39 to 41 events/revolution signals are believed to be the result of outer race movements as bearing balls pass flange bolt positions. The bearing contains 12 balls and 8 flange bolts. With a reported contact angle range calculated to be between 26.83^o to 30.05^o, bearing cage frequencies range from 43% to 44% of the rotor speed, producing the observed events/shaft revolution.

Additional forces on the bearing outer race are ball centrifugal forces and are calculated as follows:

$$F_c = \frac{(m_{ball} \times r_{pitch\ rad.})\ \omega^2}{g_c}$$

Where: F_c = centrifugal force (lb)
M_{ball} = mass of bearing ball (lb)
= 0.121 lb
$r_{pitch\ rad.}$ = bearing ball pitch radius (in)
= 3.125 in
ω = ball angular speed (rad/sec)
= $(\text{cage speed}/9.55)^2$ = 204626
g_c = gravity constant (386 in/s^2)

F_c = 200.4 lb at an engine speed of 7600 rpm and where cage speed depends on bearing contact angle. For this calculation, contact angle was assumed to be 28 degrees giving a cage speed of 55.27 Hz and a ball pass frequency of 663.24 Hz (12 times cage speed). REBAM® data indicates 3 µin to 8 µin pp deflection amplitudes in this region.

Baseline Data from Shaft Probes

Figure 6 shows steady state direct compensated and 1X filtered compensated orbit/timebase data at 7800 rpm from the two shaft probes viewing the coupling. The direct orbit/timebase has been waveform compensated to subtract the slow roll runout waveform. The 1X amplitude was recorded at 4.17 mils pp and is largely the result of rotor imbalance. The 2X component was recorded at 1.47 mils pp in both the forward and reverse directions. This 2X amplitude is also evident in the REBAM data and its source is still under investigation.

Figure 7 shows the coastdown data from both the shaft probes as well as the internal and external accelerometers. A well-damped rotor resonance is clearly evident between 4200 cpm and 5000 cpm in the proximity probe data that is not seen in the data from the center case-mounted accelerometer. The internal accelerometer shows some evidence of this same resonance, but to a lesser degree than the proximity probes.

A second resonance between 6100 cpm and 6900 cpm, predominantly evident on the accelerometers, appears to be a casing structural or system resonance. This is shown in the Bode plots of Figure 7.

Gap voltage data from the two shaft probes is shown in Figure 8. As noted earlier, the mounting position of the two probes places them viewing the edge of the shaft coupling when the unit is cold and unloaded. As the unit loads and warms, the axial position of the coupling grows toward the probes such that more of the target area is under the eddy current field and the average gap voltage goes more positive. Although unintentional, this gap voltage change can be used to track axial position of the rotor relative to the probes. Testing of the coupling material as it moves laterally under the eddy current field is still being conducted to determine a correlation between gap voltage change and axial position.

Baseline Data from Internal Accelerometer

As expected, data from the internal and external accelerometers differ considerably in frequency content and amplitude. Figure 9, cascades from the startup and coastdown, indicate the external accelerometer is registering higher-end frequency components not seen by the bearing housing-mounted internal transducer.

These higher-end frequency components, in the 45 to 55 kcpm range, are thought to be blade passing or other structural components. While the case-mounted accelerometer reflects these higher order frequency components, the overall direct amplitude of the internal accelerometer is in some cases 2 to 3 times the level of the case-mounted transducer. This is likely because of still higher order frequency components beyond the 1000 Hz sampling range in this data set and because of rotor signal attenuation through the casing.

Field data from engine

Presently, with over 15,000 operational hours since the overhaul, the engine continues to operate normally; however, recent REBAM data has shown an increase in the center bearing inner-race frequency component, as shown in Figure 10. Although still low in amplitude at 6 µin to 8 µin pp, this component has increased 200% to 300% since late last year. The REBAM probe located at 135 degrees also shows evidence of the increase in this component amplitude. Interestingly, neither the internal HTAS accelerometer, the shaft proximity probes, nor the casing accelerometers have observed this increase yet. Daily remote monitoring of the vibration data via the Data Manager® 2000 system is still taking place. The steady

state data plot formats from all transducers being used in the analysis of the engine include the following:

- Direct amplitude trends.
- 1X filtered amplitude and phase trends.
- Trended asynchronous spectrum waterfalls.
- Real-time asynchronous spectrums.
- Gap voltage trends.

Plans for the unit are to continue gas load demand operation. Current load has kept the unit in the range of 7200 rpm to 7600 rpm for the past 18 months, with several short outages. As vibration data indicates further degradation, it will be closely monitored and unit operations notified of any changes or trends. Shutdown criteria for the unit is still based on vibration alarm limits set for the two casing-mounted accelerometers. Although still preliminary, it appears that data from the internal transducers is forecasting mechanical changes well in advance of the casing transducers and will be able to predict bearing problems long before reaching alarm level.

Project Future

Items still to be completed for the Avon project are as follows:

- Post-fault overhaul to determine root cause engine problems.
- Post-fault overhaul to evaluate condition of internal transducer suite.
- Document internal vs. external transducer data comparisons.
- Establish "Best Practices" for transducers and monitoring on Avon units.
- Agreement by Rolls-Royce on a "Best Practices" vibration monitoring document.
- Publication of Results.
- Establish retrofit Avon transducer "kit."

Acknowledgments

Thanks go to the many people in Bently Nevada and TransCanada PipeLines, TransCanada Turbines, and Rolls-Royce for their assistance and cooperation without which this project would not have taken place.

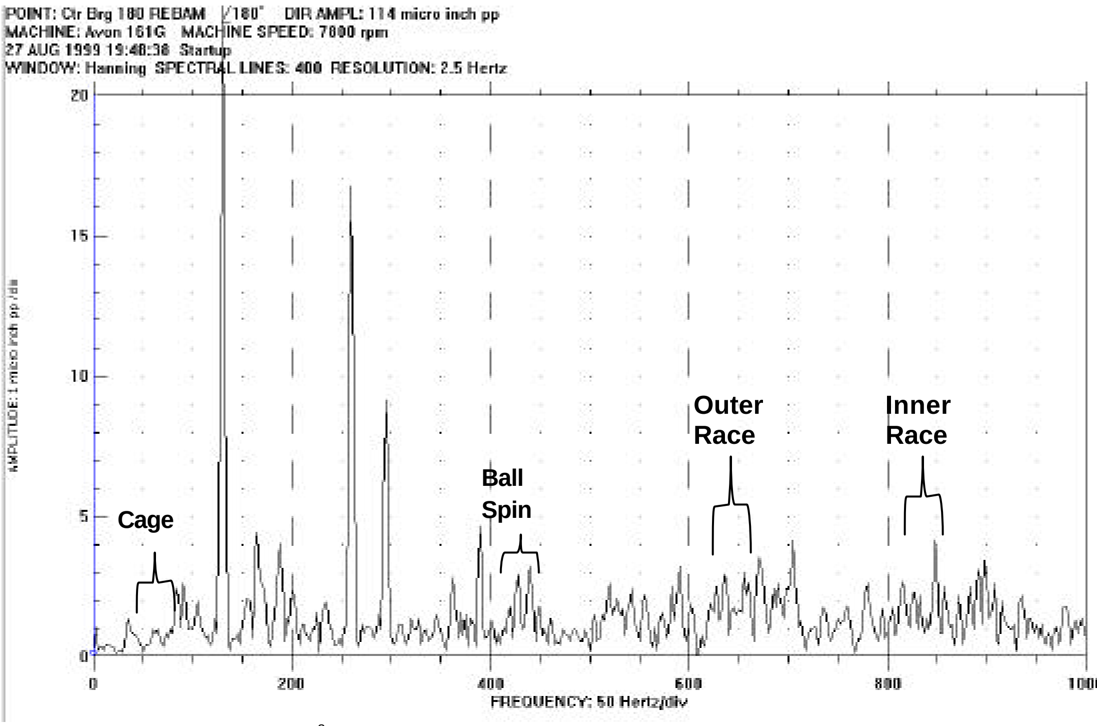


Figure 5. Baseline data from 180° REBAM probe at 7800 rpm engine speed.

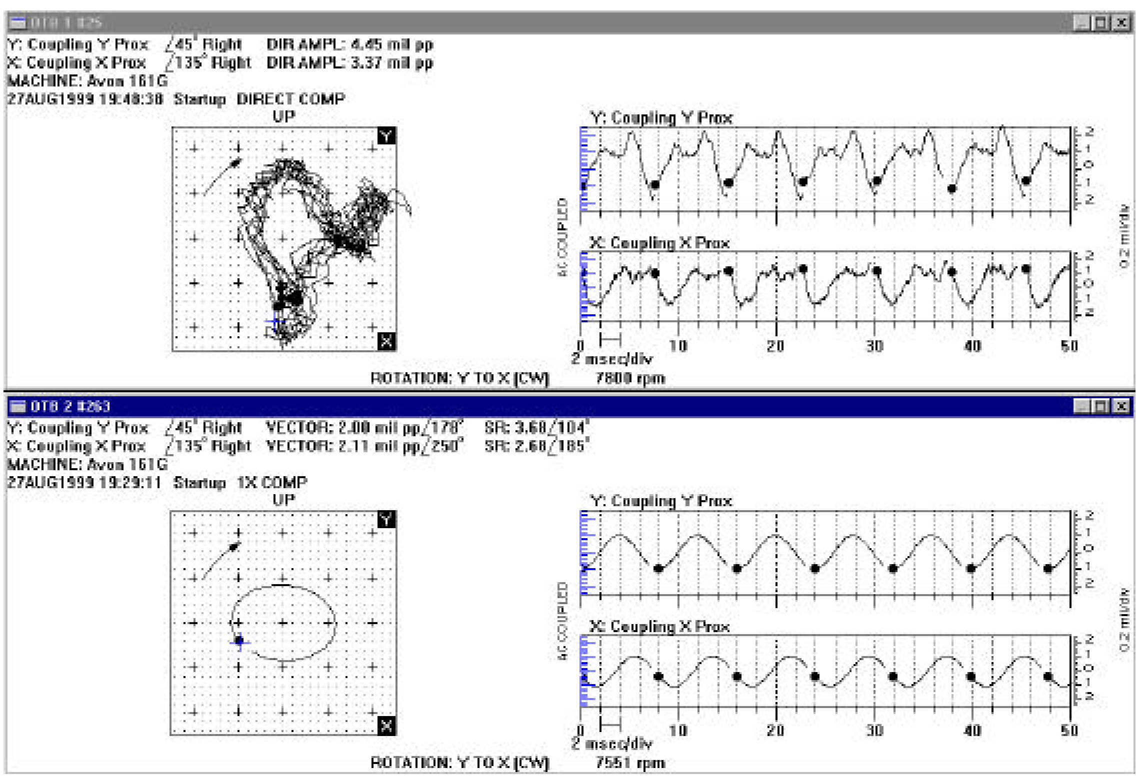


Figure 6. Baseline data for shaft probes. Top plot shows direct compensated, unfiltered orbit/timebase. Bottom plot shows 1X filtered compensated orbit/timebase.

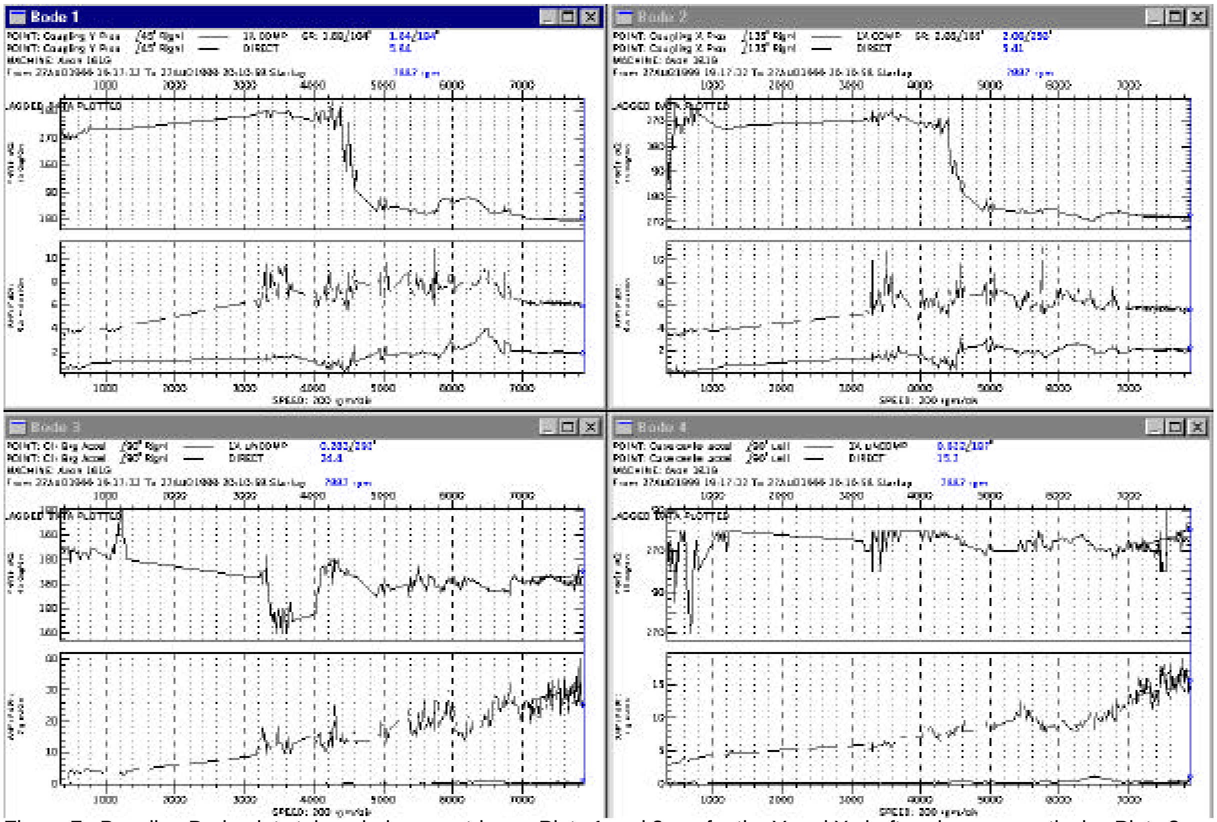


Figure 7. Baseline Bode plots taken during coastdown. Plots 1 and 2 are for the Y and X shaft probes, respectively. Plots 3 and 4 are from internal and external accelerometers, respectively.

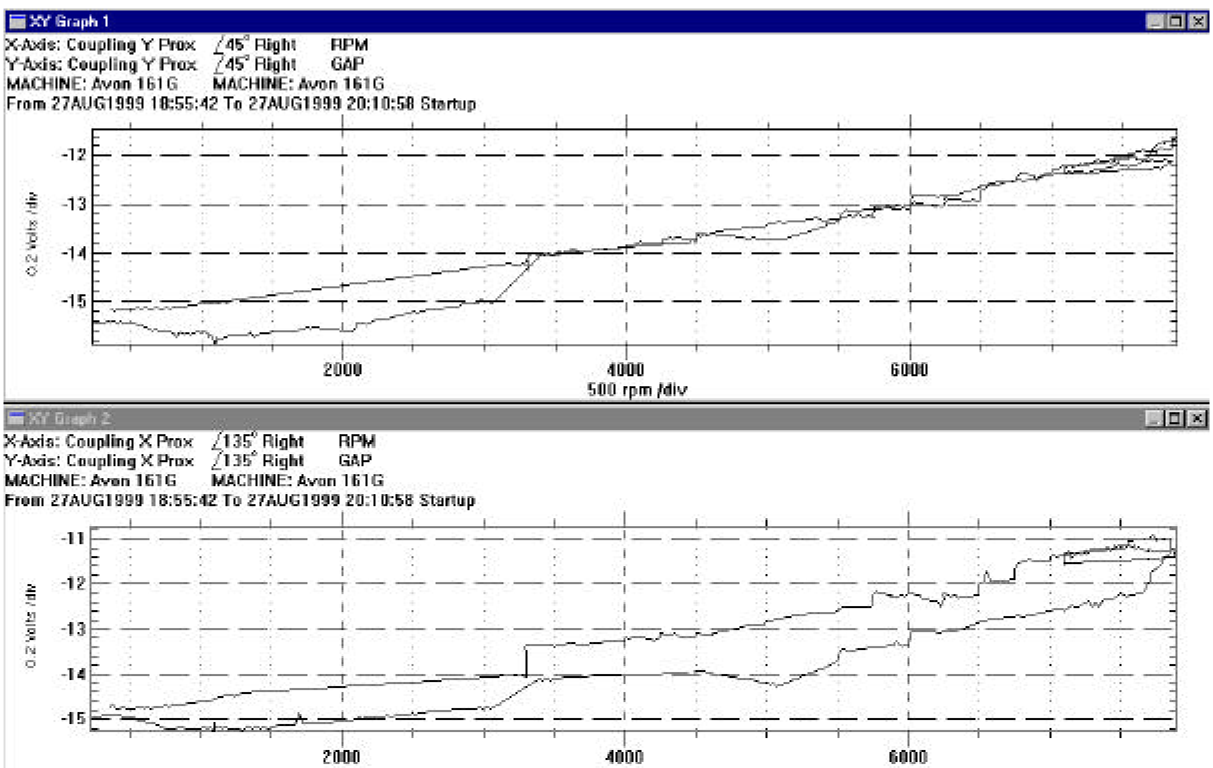


Figure 8. Gap voltage trends from shaft probes during startup, steady state, and coastdown conditions.

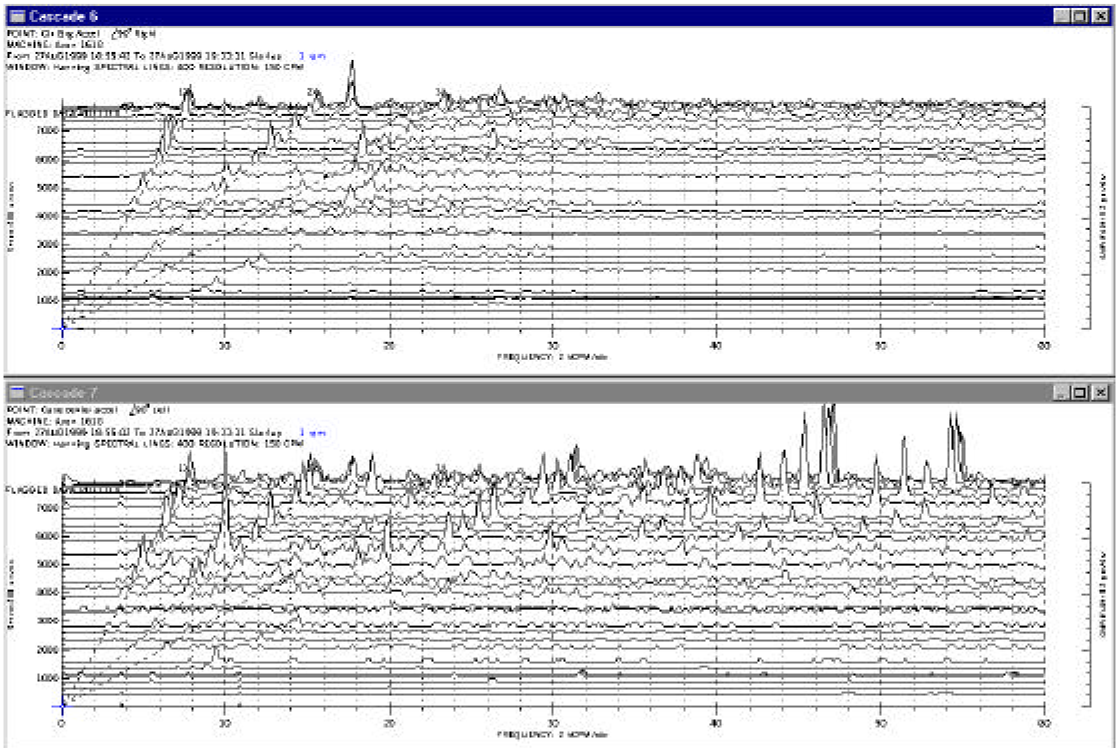


Figure 9. Spectrum cascades, using same scaling, for accelerometers. Internal accelerometer is shown on top, case-mounted accelerometer is shown on bottom.

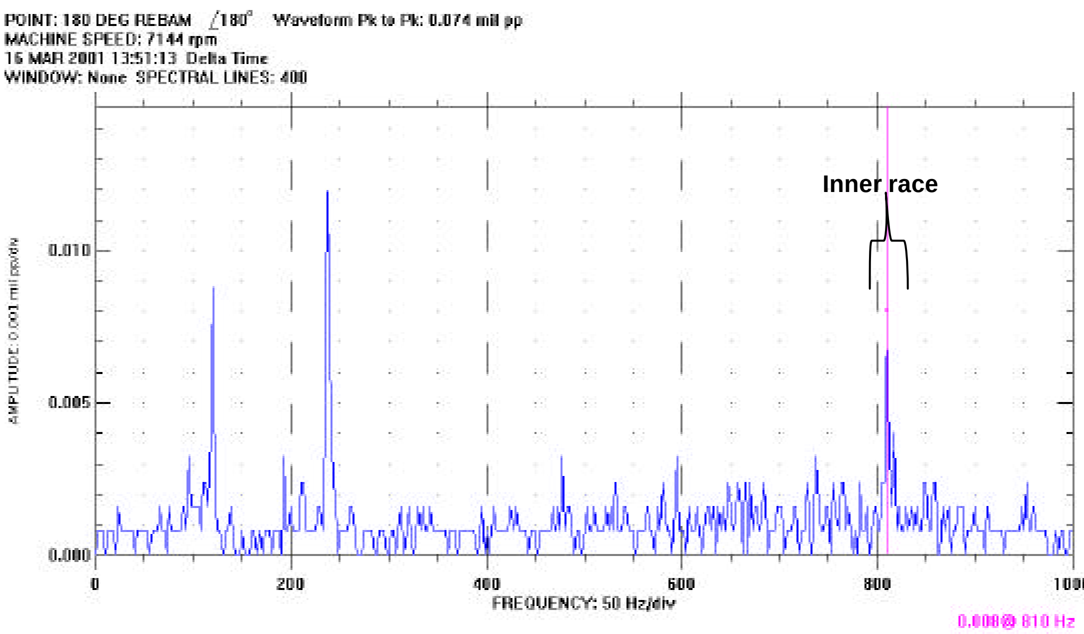


Figure 10. Spectrum of the 180° REBAM probe after engine had been in operation for 15,000 hours. Cursor position coincides with inner-race frequency. Note increase in inner-race component compared to baseline data of figure 5.

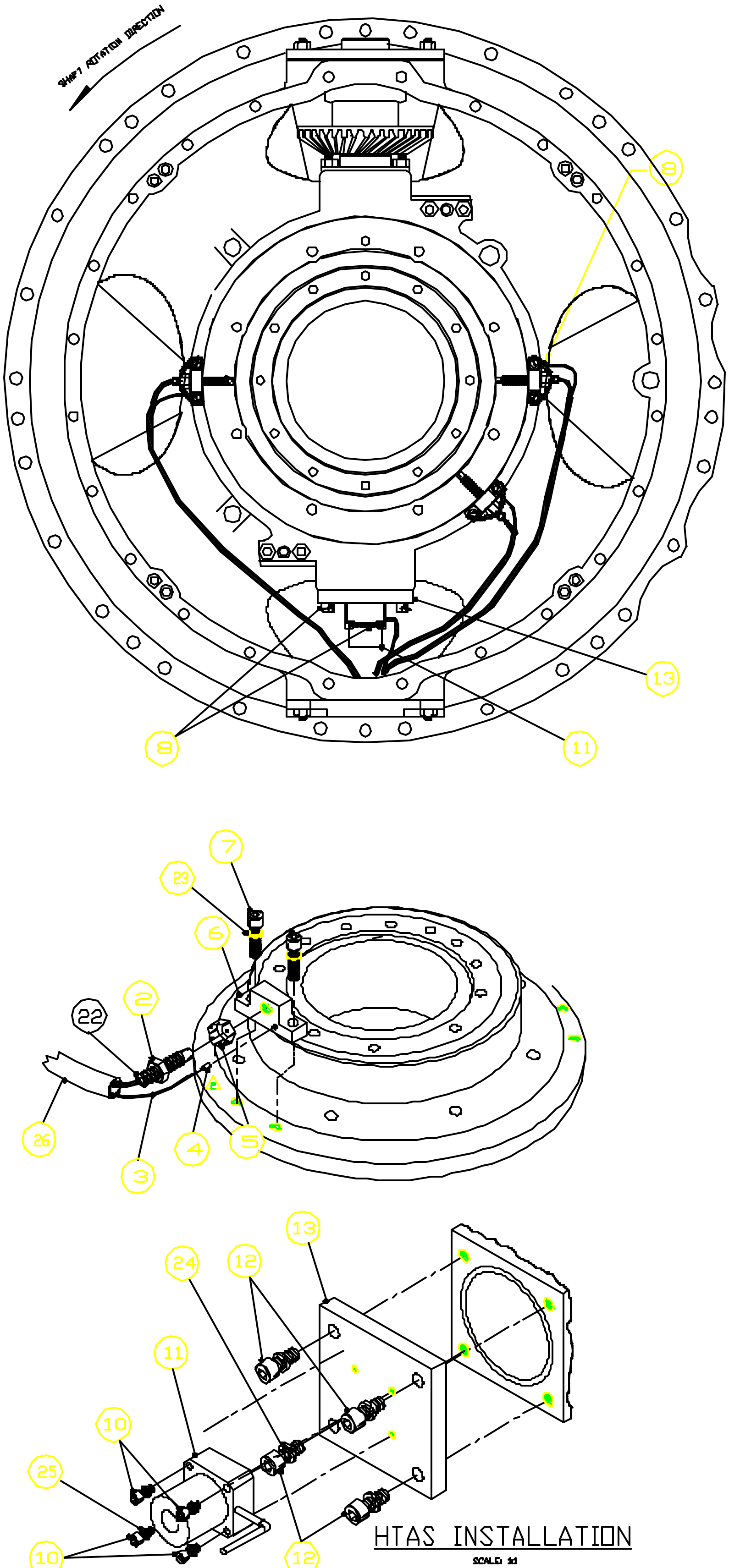
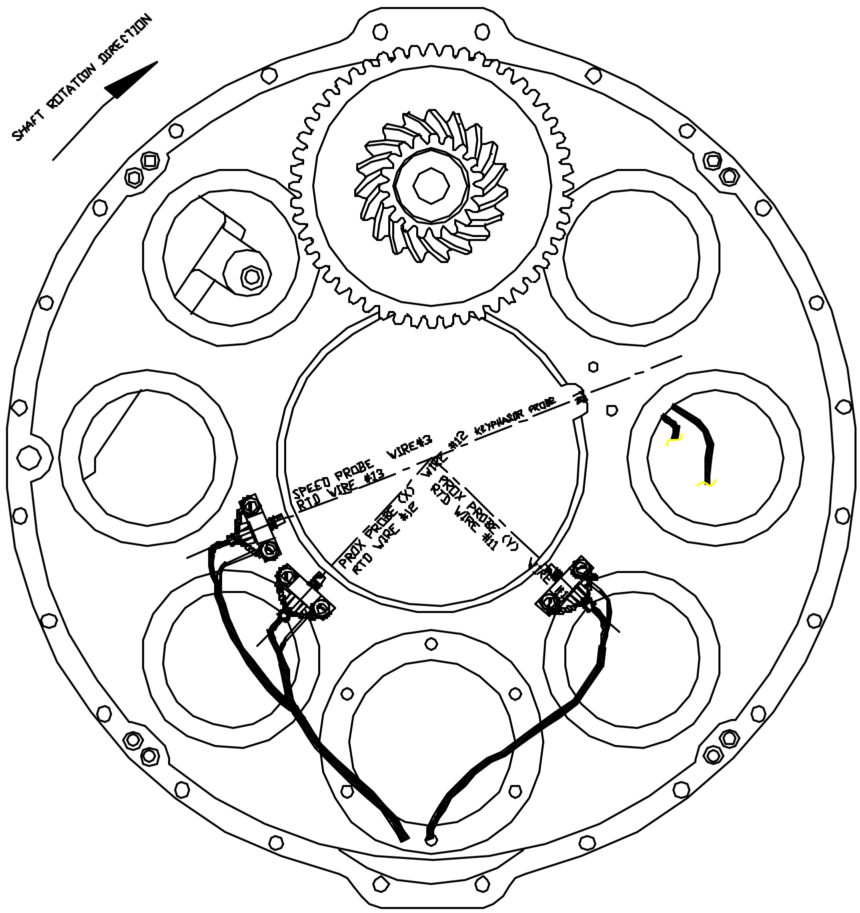


Figure 11. Front half of internal wheel case, looking forward, showing three REBAM probes and internal accelerometer mounting details.



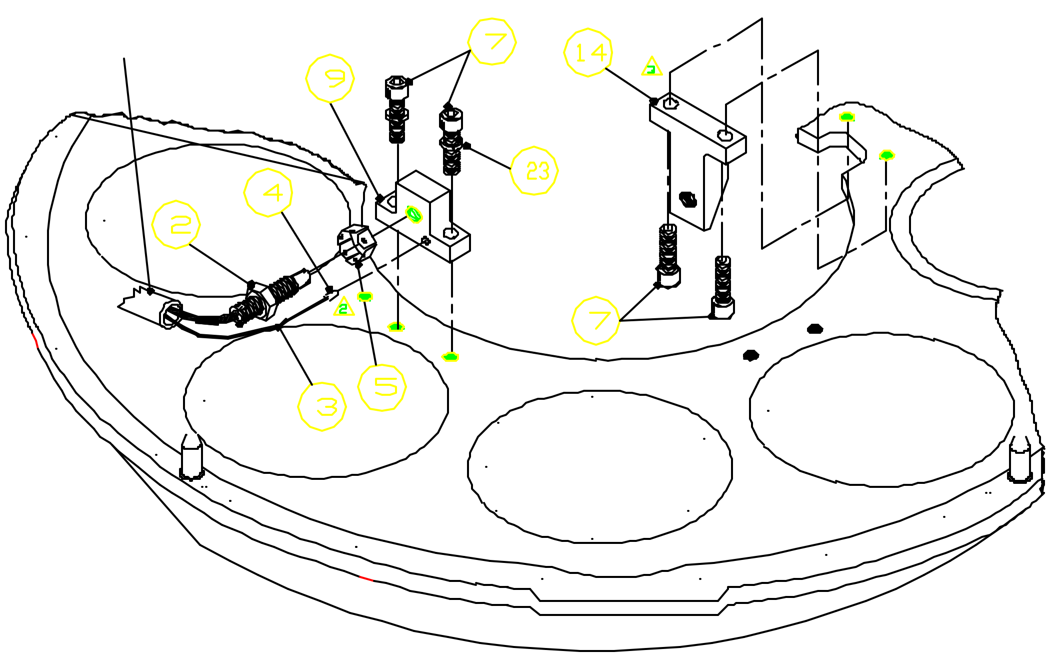


Figure 12. Rear half of internal wheel case, looking aft, showing shaft probes and speed probe (top). Bottom detail shows one shaft probe and mounting bracket for initially proposed Keyphasor® probe.

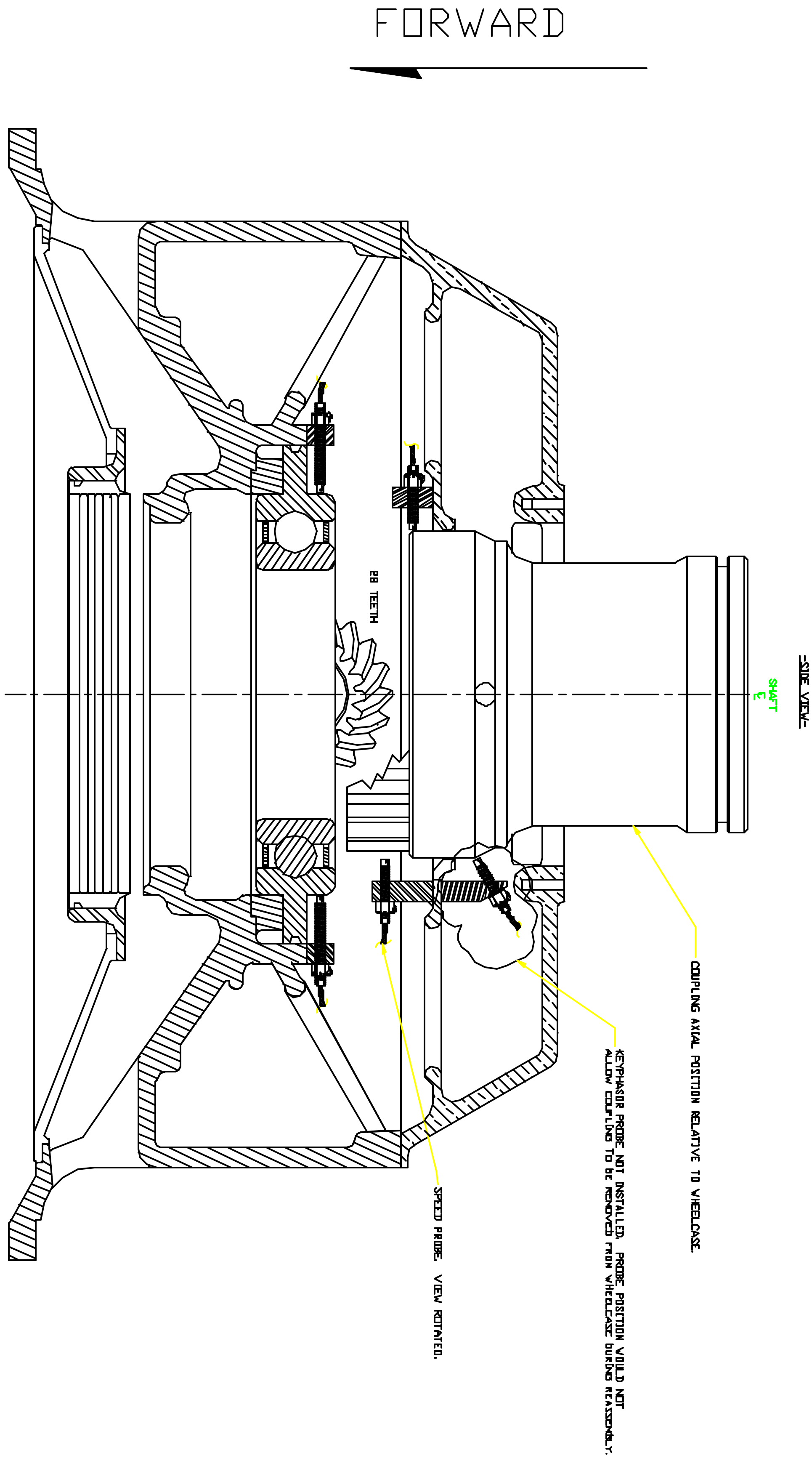


Figure 13. Cross section of internal wheel case with transducers. Two of the three installed REBAM probes are visible, one of the two shaft probes is visible, and the speed probe is visible in this view. The initially proposed Keyphasor® transducer is also shown, but an alternate location has subsequently been chosen.